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THE EFFECTS ON SWAY ANGLE PERFORMANCE IN GANTRY CRANE SYSTEM BY USING PSD ANALYSIS

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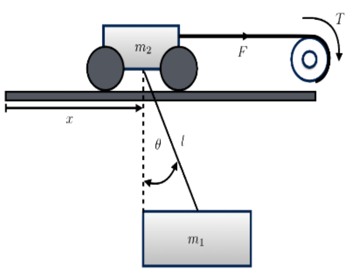
Graphical abstract



Abstract

Gantry Crane is also known as an overhead crane and widely used in industries, constructions or shipyards due to limited human capability to carry the various types of load. This system is developed to load and unload heavy materials from one place to another desired location. The problem is frequently occurs when the crane has to move the load at the required position while minimizing the sway angle of the oscillation. Thus, this research presents the investigation of the 2-D Gantry Crane System which focuses on the sway angle characteristics via Power Spectral Density (PSD) analysis. The mathematical dynamic model of the Gantry Crane System is developed using the Lagrange Equation. The system is simulated in MATLAB/Simulink environment and the results are presented in the form of time and frequency domain. A comparative assessment of the various payload mass and rope length of the system performance is presented and discussed.

Keywords: Gantry Crane System; Power Spectral Density; sway angle; bang-bang input; dynamic modeling.

Abstrak

Kren gantri yang juga dikenali sebagai kren overhed dan digunakan di dalam industri, pembinaan atau limbungan secara meluas disebabkan oleh kupayaan manusia yang terhad untuk membawa pelbagai jenis beban. Sistem ini dibangunkan bagi memuat dan memunggahkan bahan-bahan berat dari satu tempat ke tempat lain yang dikehendaki. Masalah sering berlaku ketika kren memindahkan beban pada kedudukan yang dikehendaki disamping meminimumkan sudut huyung ayunannya. Dengan ini, penyelidikan ini membentangkan kajian ke atas Sistem 2-D Sistem Kren Gantri yang memberi tumpuan kepada ciri-ciri sudut bergoyang melalui Ketumpatan Kuasa Spektrum (PSD) analisis. Model matematik dinamik menerangkan Sistem Kren Gantri telah diperolehi dengan menggunakan persamaan Lagrange. Sistem ini disimulasi dalam persekitaran MATLAB/Simulink dan hasilnya dipersembahkan dalam bentuk domain masa dan frekuensi. Satu penilaian perbandingan tahap dalam pelbagai jisim muatan dan kepanjangan tali dibentang dan dibincangkan.

Kata kunci: Sistem Kren Gantri; Ketumpatan Kuasa Spektrum; sudut bergoyang; input bang-bang; model dinamik.

1.0 INTRODUCTION

A crane is a type of machine which equipped with a hoist, wire ropes or chains and sheaves. It is used for both lifting and transporting heavy things from a place to another. Cranes are commonly employed in the construction and manufacturing industries. In the construction industry, cranes are used for loading and unloading the freights whereas in the manufacturing industry, it is used for assembling heavy equipment.

Gantry crane is important in the transportation industry for loading and unloading load [1-2]. The control objective of this system is to move the trolley to a desired position as fast and accurate as possible without causing any excessive swing at the final position [3]. In transportation industry, speed is required as the priority issue as it translates into productivity and efficiency of the system [4-5]. However, controlling the crane manually by human will tend to excite sway angles of the hoisting line and degrade the overall performance of the system. At very low speed, the payload angle is not significant and can be ignored, but not for high speed condition. The sway angle becomes larger and harder to settle down during movement and unloading [6-8]. Besides, the effect of increasing the hoisting will degrade the performance of sway angle [9-10]. This is very severe problem especially in the industries which require small of sway angle with time taken for the transportation is short and high safety [11-12].

This paper presents a mathematical modeling structure that provides a closed-form dynamic equation of motion of the 2-D Gantry Cane System. The work also presents the effects of payloads on the dynamic behavior of Gantry Crane System. The Langrage Equation principle is used to derive the dynamic model of the system. The developed model structure is implemented and simulated in MATLAB/Simulink environment. This paper only focuses on the payload sway. Power Spectral Density (PSD) is obtained in both time domain and frequency domain. The study of the effect of payload sway is presented with varying the parameters of the Gantry Crane System.

2.0 DYNAMICS MODEL STRUCTURE

The system includes two masses, m_1 and m_2 which represented payload mass and trolley mass respectively. The cable length used for supporting the payload mass. The system in a 2-D system which in x-axis and y-axis coordinates only. The x-axis coordinated as movement of the trolley horizontally and y-axis coordinated as for the swaying of the payload mass.

In this section, a dynamic model of nonlinear gantry crane is formed. Assuming the dynamic model has the characteristics that the trolley and payload are connected by a massless, rigid link as in Figure 1 and the parameters of the system is shown in Table 1.

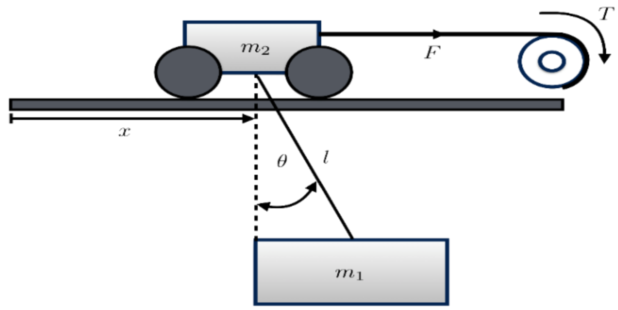


Figure 1 Dynamic model structure of gantry crane [12]

Table 1 Dynamic model structure of gantry crane [13]

Parameters	Unit	Values
Payload mass	m_1	0.5 to 1.5kg
Trolley mass	m_2	5.0 kg
Rope length	l	0.5 to 1.5 m
Gravitational	g	9.81ms^{-2}
Friction	B	12.32 Nsm^{-1}
Resistance	R	2.6 Ω
Torque constant (K_t)	K_t	0.007 NmA^{-1}
Electric constant (K_e)	K_e	0.007 Vsrad^{-1}
Radius of pulley (r_p)	r_p	0.02 m
Gear ratio (z)	z	0.15

From the previous study [14-16], it has shown that Langrage Equation is frequently used to derive the model of the Gantry Crane System. Therefore, Langrage Equation is chosen for mathematical modeling of the system.

$$L = T - P \tag{1}$$

where:

L : Lagragian function
T : Kinetic energy
P : Potential energy

The kinetic energy in Equation (2) and potential energy in Equation (3) of the whole system are:

$$T = \frac{1}{2}m_1\left(\dot{x}^2 + l^2\dot{\theta}^2 + 2\dot{x}l\dot{\theta}\cos\theta\right) + \frac{1}{2}m_2\dot{x}^2 \tag{2}$$

$$P = -mgl\cos\theta \tag{3}$$

Constructing Equation (2) and (3) by using Lagrange Equation as in Equation (4).

$$\frac{d}{dt}\left(\frac{\partial L}{\partial \dot{q}_i}\right) - \frac{\partial L}{\partial q_i} = Q_i \qquad (4)$$

where:
- q_i : Independent generalized coordinate
- Q_i : Nonconservative generalized coordinates

The following equations of motion can be obtained as:

$$(m_1 + m_2)\ddot{x} + m_1 l\ddot{\theta}\cos\theta + 2\dot{x}l\dot{\theta}\cos\theta + B\dot{x} - m_2 l\dot{\theta}^2 \sin\theta = F \qquad (5)$$

$$m_1 l\ddot{x}\cos\theta + m_1 l^2\ddot{\theta} + m_1 gl\sin\theta = 0 \qquad (6)$$

The force in the Equation (5) derived from the DC motor as in Figure 2.

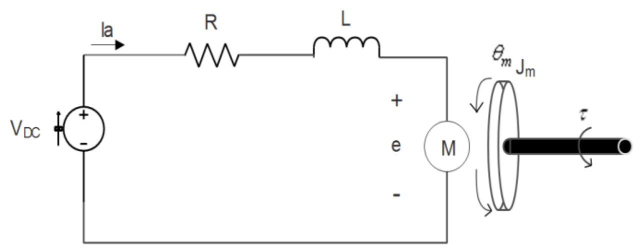


Figure 2 **DC** motor circuit diagram

By using Kirchhoff Current Law in Figure 2, Equation (7) is derived.

$$V = RI_a + L\frac{di}{dt} + V_b \qquad (7)$$

Where L is neglected because the value is very small. The equation became as Equation (8).

$$V = RI_a + V_b \qquad (8)$$

By applying Newton's 2nd Law of motion to the motor shaft [17], Equation (9) is obtained.

$$J_m\theta_m = T_m - \frac{T_L}{r} \qquad (9)$$

Since the moment of inertia, J_m is very small, then the equation become as in Equation (10):

$$T_m = \frac{T_L}{r} \qquad (10)$$

where:
- V : Voltage
- R : Resistance
- I_a : Armature current
- L : Inductance
- T_m : Motor torque
- T_L : Load torque

From Equation (7) to (10), it derived force, F in Equation (11) at the trolley in x-axis.

$$F = \frac{VK_t z}{Rr_p} - \frac{K_e K_t z^2}{Rr_p{}^2}\dot{x} \qquad (11)$$

Thus, the differential equation for the system are:

$$(m_1 + m_2)\ddot{x} + m_1 l\ddot{\theta}\cos\theta + B\dot{x} - m_1 l\dot{\theta}^2 \sin\theta = \frac{VK_t z}{Rr_p} - \frac{K_e K_t z^2}{Rr_p{}^2}\dot{x} \qquad (12)$$

$$m_1 l\ddot{x}\cos\theta + m_1 l^2\ddot{\theta} + m_1 gl\sin\theta = 0 \qquad (13)$$

The equation of the position and sway in the overall system is in the Equation (14) and (15). In this paper, only sway part will be focused.

$$\ddot{x} = \left(\begin{array}{l}\frac{VK_t z}{Rr_p} - \frac{K_e K_t z^2}{Rr_p{}^2}\dot{x} - B\dot{x} - \\ m_1 l\ddot{\theta}\cos\theta + m_1 l\dot{\theta}^2 \sin\theta\end{array}\right)\left(\frac{1}{m_1 + m_2}\right) \qquad (14)$$

$$\ddot{\theta} = \left(\frac{1}{m_1 l^2}\right)(m_1 l\ddot{x}\cos\theta + m_1 gl\sin\theta) \qquad (15)$$

3.0 SIMULATION

Figure 3 shows the structure of Gantry Crane System. Meanwhile, Figure 4 shows the design with the development of position and sways in Equation (8) and (9) respectively in MATLAB/Simulink with the bang-bang force input [18-20].

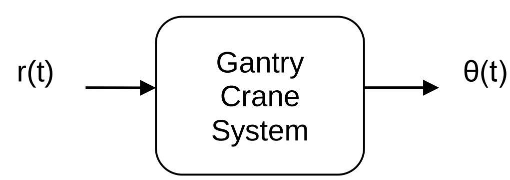


Figure 3 Gantry Crane System structure

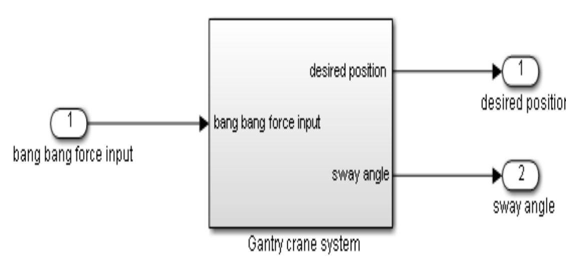


Figure 4 The Gantry Crane System block model

Figure 5 illustrates the subsystem build in the Gantry Crane System block as in Figure 4. Lastly, the simulated block diagram in the Simulink platform is shown in the Figure 6.

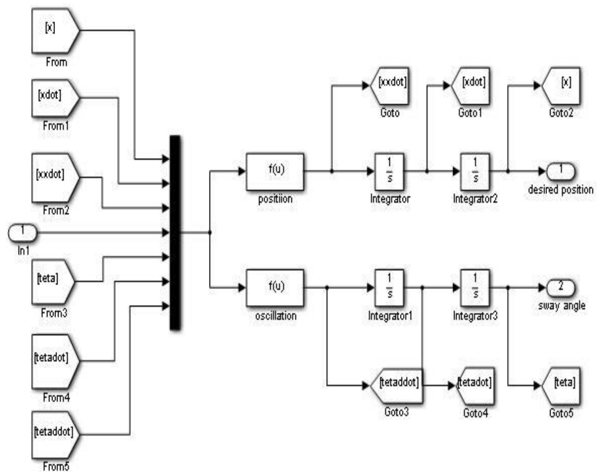


Figure 5 Subsystem in the Gantry Crane System [12]

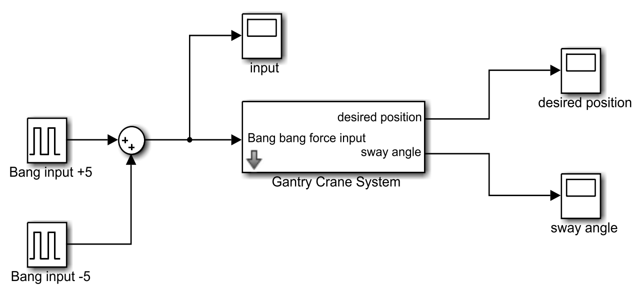


Figure 6 Block diagram of Gantry Crane System simulated in Simulink

Figure 7 shows a bang-bang force input injected into the system. This input is plotted in positive and negative value in one cycle or period so that the system able to move the trolley to accelerate (positive) and decelerate (negative) and possible to stop (zero) [21] at the desired position and the sway angle is minimized.

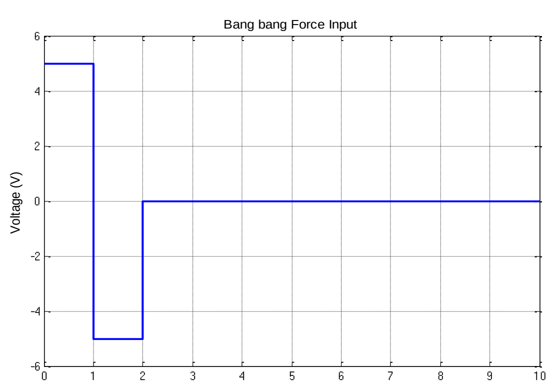


Figure 7 Bang-bang force input

4.0 RESULTS AND DISCUSSION

The sway angle produced from the payload mass in Figure 8 illustrates the sway angle of the system decreased as the load increasing. In other words, the behavior of the sway angle is increasing as the weight is increased. When the payload mass is 0.5 kg, the maximum sway is obtained in the system is 7.0368×10^{-3} rad and time per cycle is 1.6539 s. Therefore, the frequency created in the system when handling this mass is 0.6046 Hz as f = 1/T. When the payload mass is 1.0 kg, the maximum sway obtained in the system is 6.462×10^{-3} rad with time per cycle is 1.5852 s which is 0.6308 Hz. Then, the payload mass is increased to 1.5 kg. The maximum sway obtained at this point is 5.8273×10^{-3} rad with 1.5258 s per cycle and frequency of 0.6554 Hz. Table 2 shows summarize of response of payload sway in time domain of various payload mass.

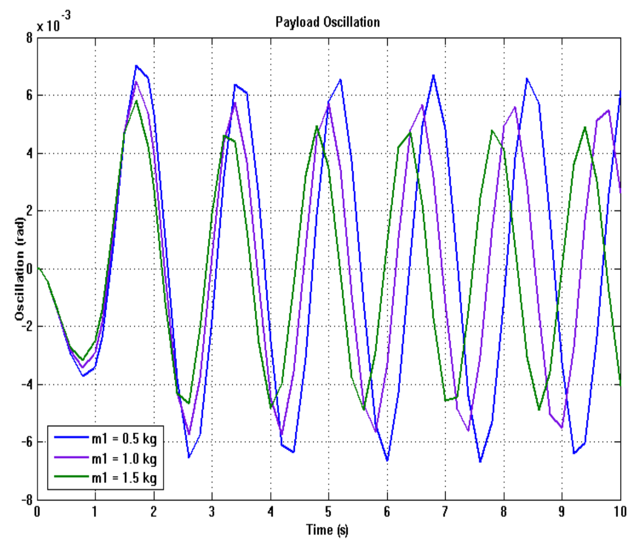


Figure 8 Response in time domain of various of payload mass (m_1)

Table 2 Summarize of response in time domain of various of payload mass (m_1)

Payload mass, m_1 (kg)	Payload oscillation, θ_{max}, (rad)	Period, T (s)
0.5	7.0368×10^{-3}	1.6539
1.0	6.4623×10^{-3}	1.5852
1.5	5.8273×10^{-3}	1.5258

The performances of the sway angle were analyzed in Power Spectral Density (PSD). PSD is a measure of a signal power intensity in the frequency domain to identify oscillatory signals in time series data and verify their amplitude. It is also states at which frequency ranges variation is strong and that might be quite useful for further analysis. There were three sets of mass had been tested in payload section which were 0.5 kg, 1.0

kg and 1.5 kg. The performance of the sway is analyzed in PSD. Based on Figure 9, when the payload mass is 0.5 kg, the PSD is at the normalized frequency of 0.2344 π rad/sample. When the mass is tested with 1.0 kg, the value of normalized frequency in PSD analysis is at 0.2500 π rad/sample which is higher than 0.5 kg. Then, the test is continued with 1.5 kg of payload mass. It shows that the frequency created at 0.2578 π rad/sample. Table 3 shows the summarized of the response in the frequency domain with various of payload mass.

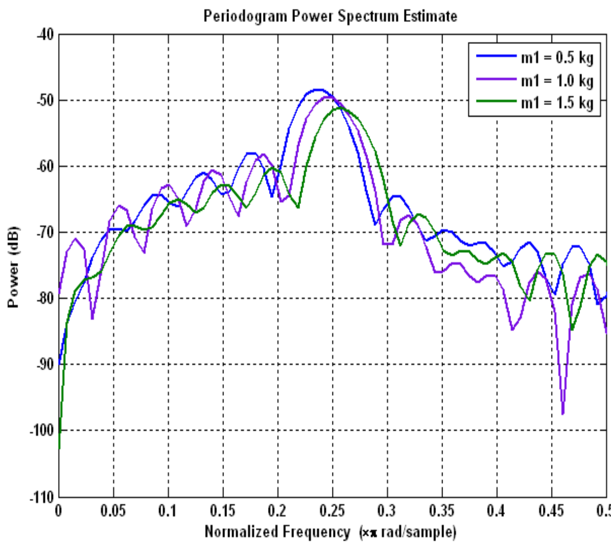


Figure 9 Response in frequency domain of various of payload mass (m_1)

Table 3 Summarize of response in frequency domain of various of payload mass (m_1)

Payload mass, m_1 (kg)	Power, (dB)	Normalized frequency, (π rad/sample)
0.5	-48.41	0.2344
1.0	-49.75	0.2500
1.5	-51.23	0.2578

Figure 10 shows that with the rope length of 0.5 m hanging the payload mass with 0.5 kg, the maximum sway obtained is 6.1427×10^{-3} rad. The time taken for a cycle is 1.3516 s with equal to 0.739 Hz. When the length is increased to 1.0 m, the sway performance shown that 7.4693×10^{-3} rad is the maximum sway and the time taken for a complete cycle is 1.9126 s with a frequency of 0.5228 Hz. Then, the length is increased again to 1.5 m. The maximum sway obtained is 7.0845×10^{-3} rad with time taken per cycle is 2.3435 s and frequency is 0.4267 Hz. The summarize of the response of payload sway in time domain with various of rope length is shown as in Table 4.

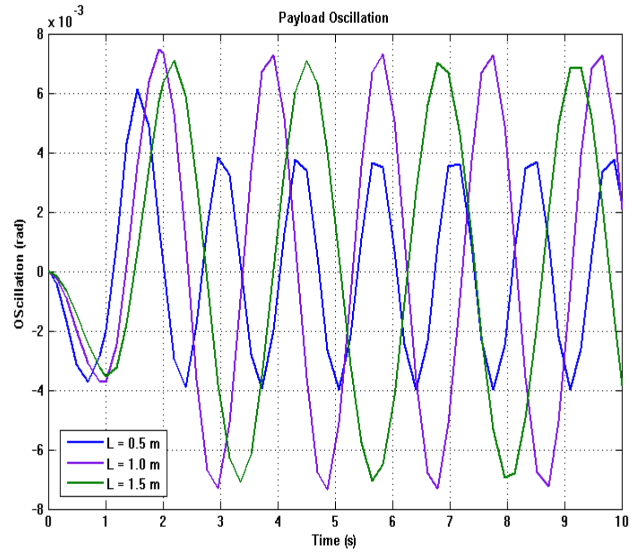


Figure 10 Response in time domain of various of rope length (*l*)

Table 4 Summarize of response in time domain of various of rope length (*l*)

Rope length, *l* (m)	Payload oscillation, θ_{max}, (rad)	Period, T (s)
0.5	6.1427×10^{-3}	1.3516
1.0	7.4693×10^{-3}	1.9126
1.5	7.0845×10^{-3}	2.3435

Figure 11 shows that when the payload mass is 0.5 m, the PSD shows the normalized frequency of 0.2734 π rad/sample which is lower than 0.5 m. Lastly, when the system is tested to 1.5 m of rope length, the frequency is created at 0.1563 π rad/sample. Table 5 shows the summarized of the response in the frequency domain with various of rope length.

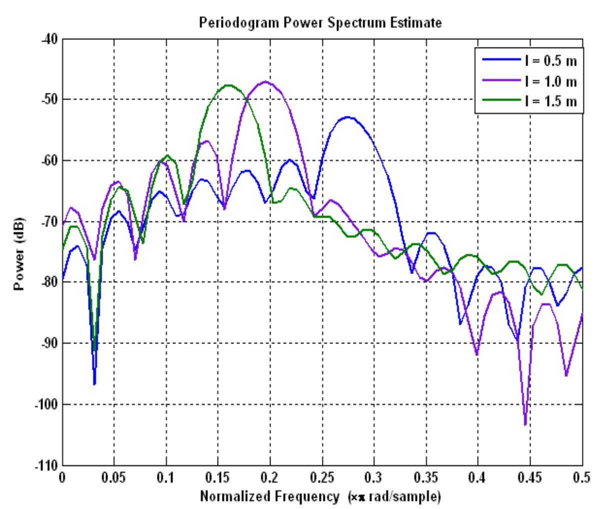


Figure 11 Response in frequency domain of various of rope length (*l*)

Table 5 Summarize of response in frequency domain of various of rope length (*l*)

Rope length, *l* (m)	Power, (dB)	Normalized frequency, (π rad/sample)
0.5	-52.89	0.2734
1.0	-47.08	0.1953
1.5	-47.78	0.1563

4.0 CONCLUSION

Investigation of the sway angle characteristics of a Gantry Crane System with a variation of payload mass and rope length has been presented. The dynamic model structure of the Gantry Crane System is proven by the Lagrange Equation that has been simulated with bang-bang input as a force. The system response in terms of sway angle has been obtained and analyzed in both time and frequency domain. The results shown that as the payload mass increased, the frequency was increased and the power in PSD analysis was decreased. Furthermore, the value of the frequency was decreased while the power in PSD analysis was increased when the rope length increased. Through these findings, it helps the researchers to improve the development of controller for the Gantry Crane System.

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